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AN INVESTIGATION OF A TURBINE CENTRIFUGAL PUMP

BY

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THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

WILLIAM WADDELL KAUTZ

ENTITLED AN INVESTIGATION OF A TURBINE CENTRIFUGAL PUMP

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE

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INVESTIGATION OF A TURBINE CENTRIFUGAL PUMP.

INTRODUCTION.

The actual results obtainable from centrifugal pumps differ considerably from the results which should obtain according to theory. This is due to incorrect assumptions as to flow thru the machine, and to indeterminate losses in the machine. Designers who have overcome the difficulties have apparently tried to keep their information to themselves for their own pecuniary advantage. Up to the present time, few of the practical methods of increasing centrifugal pump efficiency, or bettering their operation, have become very widely known, and a number of tests must be run on any design of pump in order to determine its actual performance.

Within the last few years, centrifugal pumps have come into use for fire service, city pumping, boiler feed, etc. The small floor space they occupy is the greatest point in their favor, although their adaptability to pumping large quantities of water against a low head is no slight advantage.

The object of this thesis is to investigate the operation of a turbine type centrifugal pump. The pump under test in this investigation is a two stage, turbine type centrifugal pump, manufactured by Henry R. Worthington Co., which is owned by the department of Theoretical and Applied Mechanics of the University of Illinois.

METHOD OF TESTING:

The test of the pump investigated was made to deter-

mine the range of operation and the corresponding efficiencies for the pump. For each run the speed was kept constant and the discharge and efficiency determined for various heads. The pump was placed over the sump in the Hydraulic Laboratory of the University of Illinois. For most of the runs the water was pumped through 80 feet of 12 inch pipe and discharged into a weir channel. In a few runs, however, the water was pumped through 110 feet of 8 inch pipe and discharged into a weir box. In all runs the head was varied by throttling the discharge valve, and the water was measured over a weir, then allowed to return to the sump to be used over again.

APPARATUS: The pump was belt-connected to a 100 horse power Ideal Engine. As stated before, the pump is a two stage centrifugal pump of the turbine type. The runners are 15 inches in diameter and are made up of two circular discs with the runner blades between them, all cast in one piece. The guide vanes surround the runner and their outside diameter is 28 inches. Figure 1 shows a section perpendicular to the shaft through the runner and guide vanes. Figure 2 is a vertical section through the pump showing the path of the water from suction to discharge pipe. Water enters the first runner at the center, leaves at the periphery, entering and passing thru the guide passages; it flows down from the first guide passages to the inside of the second runner, and discharges from them, through the second guide passages into the spiral trumpet shaped casing and out at the discharge pipe. There are eight vanes in each runner, four of them extending from the outside to the center

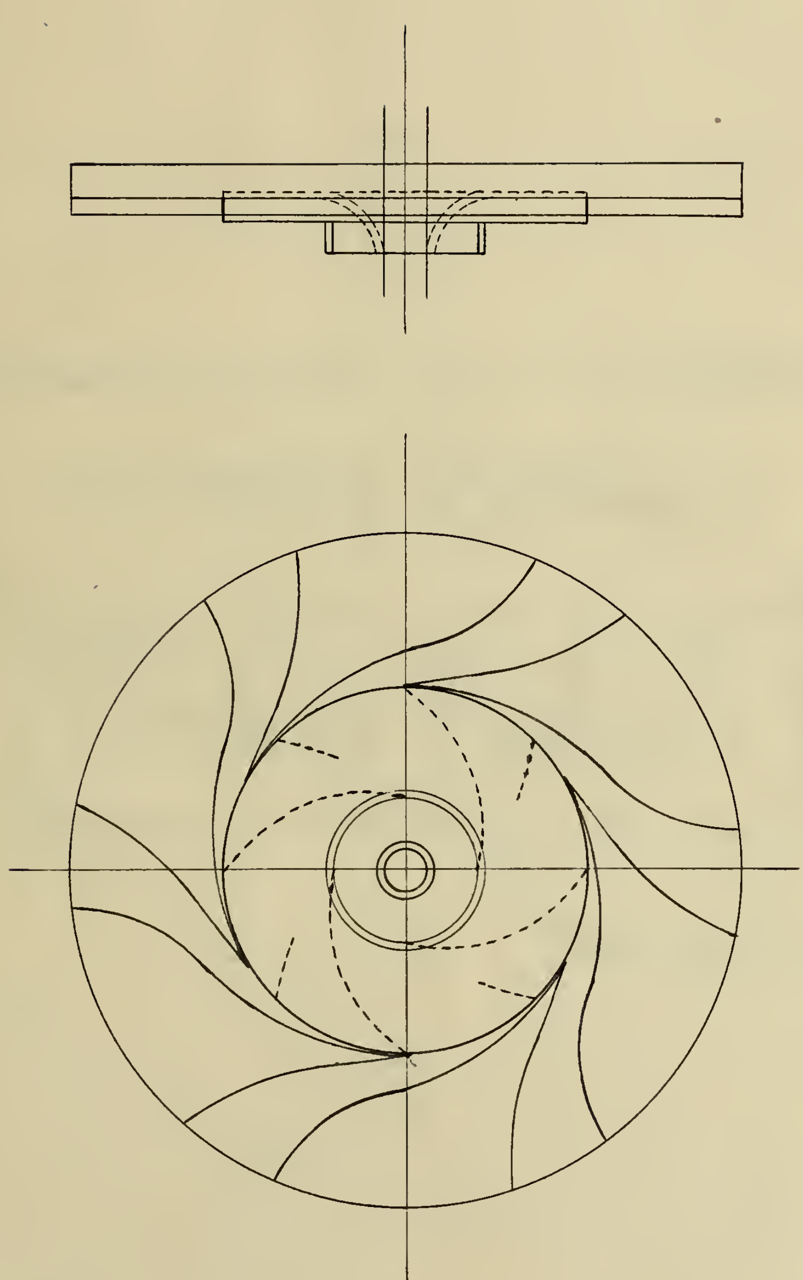


Fig 1.

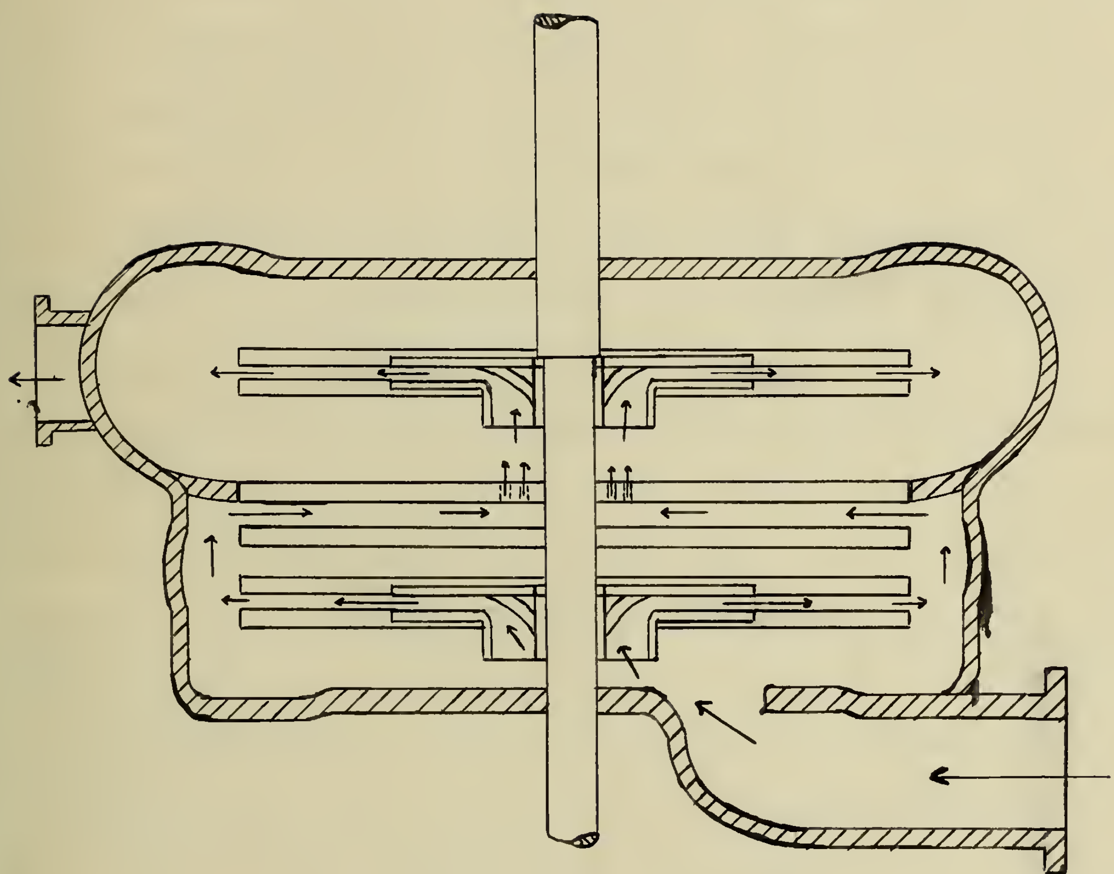


Fig. 2.

and the other four extending only half way. The direction at entrance is practically radial, and at exit the angle is twelve degrees off the tangent.

A Crosby pressure gage was placed above the pump in the discharge pipe at 45° to the axis of the pump, to indicate the head at that point. A Crosby vacuum gage placed in the suction pipe showed the suction head. Calibration curves of these gages are mot shown as they were calibrated by the department and found to be correct. Two Crosby, inside spring, steam indicators were used and cards were taken simultaneously from both ends of the engine cylinder. The springs were calibrated and found to be sufficiently correct.

The water was discharged into a weir box and was measured over a contracted weir. For some of the runs an 18 inch weir was used and for the remainder a 3 foot weir was used. The head on the weir was measured with a hook gage placed at the side of the box and about four feet back from the weir crest. Calibration curves for the two weirs are shown on curve sheets I and II. By noting the head on the crest of the weir, the discharge may be read directly from the curves. The curves were plotted between head and discharge, using values calculated by Smith's formula, using Smith's coefficients.

OBSERVATIONS: The engine was started and run at slow speed until the pump was primed, and then the speed increased until the desired R.P.M. was attained. The pump was primed by filling from a standpipe in the Hydraulic Laboratory. In each run the speed was kept constant and the head was varied; and the

discharge measured for each change of head. Readings of the head on the pump, the suction lift, the speed of engine and pump, and the head on the weir, also indicator cards, were taken every five (5) minutes. The head on the pump at starting, was made low enough that the pump operated at less than maximum economy; four or five readings were taken with the head constant, and then it was increased by (10) or (20) feet, and four or five readings taken at that head and this repeated until the maximum head was reached.

Water was pumped from the sump, through the twelve inch main, to the weir box and flowed over the weir back into the sump and the water level in the sump was noted as soon as the pipes and the weir box were filled, after which it remained constant. Water was admitted from the standpipe so that the distance from the gage on the discharge pipe to the water was constant at ten (10) feet. The speed of the engine and pump, were taken with a Starrett speed counter and a stop watch. Speed readings were taken on both engine and pump to determine the amount of slip in the belt. The head on the weir crest was read by means of a hook gage, taking a zero reading when the water level was just up to the crest, and subtracting that from the readings when the water was flowing, to get the actual head on the crest.

Indicator cards were taken from both ends of the engine cylinder, by means of the two Crosby indicators, using (40) pound springs. All readings were taken simultaneously and the head was increased in each test until the point of maximum pump economy was well passed.

The belt was thrown off and the engine run light to determine the friction of the engine parts at various speeds. Indicator cards were taken during this run with 20 pound springs in the indicators. A curve of friction horsepower is shown on curve sheet III.

TABLES:

The tables on the following pages contain the calculated data. On the test data sheets, column 1 is the number of the reading; column 2, the discharge gage reading; column 3, the distance from the discharge gage to the sump water level; column 4, the total lift, being the sum of columns 2 and 3; column 5 is the calculated discharge, in cubic feet per second; and column 6, the calculated discharge in gallons per minute; column 7 is the R.P.M. of the pump; column 8 is the net horsepower delivered to the pump, and column 9, the average net porsepower delivered to the pump at each head; column 10 is the hydraulic horsepower of the pump; column 11, the calculated efficiencies of the pump, and column 12, the average efficiency of the pump for each head.

Table No. 8 , p. 23 , shows the revolutions of the pump required to give certain discharges against different heads. It was calculated from the formula $n = K\sqrt{h + a}$ which is explained later.

RESULTS AND CONCLUSIONS:

The discharge from the pump, column 5, was obtained from the calibration curve for the weir used, curve sheet No. II ; the discharge in gallons per minute, column 6, being calculated

by the formula;

$$\text{Gallons per minute} = w \times Q \times G \times 60 \qquad (1)$$

w, being the pounds of water per cubic foot (=62.5), Q, the discharge, cubic feet per second, and G the gallons per cubic foot (=7.48).

The hydraulic horsepower of the pump, column 10, is calculated from the formula:

$$\text{Hydraulic H.P.} = \frac{Q \times w \times H \times 60}{33000} \qquad (2)$$

Q and w being the same as in (1) and H being the total head, column 4. The net horsepower delivered to the pump, column 8, is the indicated horsepower of the engine, less the friction horsepower. The engine horsepowers are calculated from the formula:

$$\text{I. H. P.} = \frac{P\ L\ A\ N}{33000}$$

P being the mean effective pressure on thepiston (the average from the two cards) pounds per square inch, L is the length of the stroke, in feet, A the area of the piston in square inches and N, the number of strokes per minute (= R. P. M. x 2).

The pump efficiencies are calculated from the formula.

$$= \frac{\text{Hydraulic H. P.}}{\text{H.P. deliv. to pump.}}$$

From the data of the tests curve sheet IV is plotted, expressing the relation between R. P. M. and head on the pump, for various discharges. Curve sheet V shows the relation between R.P.M. and discharge; curve sheet VI, the relation between R.P.M. and

efficiency and curve sheet VII shows the relation between discharge and efficiency, for various heads. From a study of the results it is evident that there is a fairly wide range of discharge for any head without much variation in efficiency, and obviously the pump may operate with the same economy under different heads. It is also evident that at any discharge the head may be varied considerably without a great change in efficiency but the best efficiencies are obtained at high heads. The reasons for this appear when the variations in the term a are noted in the formula $n = K\sqrt{h + a}$.

The maximum speed for the engine was 300 R.P.M., it being equipped with an inertia governor, which prevented exceeding that speed; this would run the pump at 1300 R.P.M., but a run could not be made with pump speed exceeding 1200 R.P.M., on account of belt slippage.

Variation of the engine indicated horsepower is perhaps mostly due to the poor quality of steam obtainable, but no doubt the average of the several cards, for each reading at constant speed and constant head, represent a true average. The friction horsepower curve is subject to a small error, although repeated trials were made to get it, and considerable care was used in taking data for it.

The losses entering into the pump operation due to various causes, are: (a) a loss due to journal friction of rotating parts, (b) a loss due to friction of the water in the suction pipe, (c) internal friction of the pump, due to friction of the water over the rotating parts of the pump; and (d), im-

pact losses, due to impact on the runner vanes. The velocity head $\frac{(V_1^2)}{2g}$ in the suction pipe requires energy and that velocity head counts as a loss against the pump. The friction is perhaps excessive through the small section in the runner, where the water enters, since the velocity must be necessarily high, when large volumes of water are discharging. There will be a loss, due to eddies, if the water has much of a velocity of whirl, at entrance to the runner.

The Worthington Company rate this pump at 450 gallons per minute against a 135 foot head, running 1500 R.P.M., this requiring about 32 H.P. This gives as pump efficiency of about (48) per cent. Much better economy was obtained at lower speeds and higher heads, as is seen from the data, and it is probable that if a speed of 1500 R.P.M. could have been reached, the pump would have operated at an economy, better than fifty (50) per cent, and against heads above two hundred (200) feet.

A close approximation to the actual loss in the machine can be arrived at by means of the following analysis:

(a) Velocity head in suction pipe

$= h_s = \frac{V_1^2}{2g}$, where v_1 is the velocity in the suction pipe, calculated from the discharge and area of cross-section: $(\frac{Q}{a_1})$.

(b) Loss at entrance to suction pipe $= .93 \frac{v_1^2}{2g}$ and this loss may be taken as $\frac{v_1^2}{2g}$, since there is a foot valve on the suction pipe.

(c) The friction loss may be calculated from the general formula for friction in iron pipes, which is:

$$h_f = f_1 \frac{l_1}{d_1} \cdot \frac{v_1^2}{2g}$$

(d) The velocity head in the discharge pipe $= h_d = \frac{v_2^2}{2g}$, where v_2 is the velocity in the discharge pipe, calculated from $\frac{Q}{a_2}$.

(e) Another loss of head occurring is what might be termed the "machine loss", which is that due to friction, impact, etc. inside the casing. ¶ The R.P.M. required to give a certain discharge is given approximately by the following equation:

$$n = K\sqrt{h + a},$$

where "n" is the R.P.M. of the pump, "h" is the total head, represented by the sum of the gage reading and the distance from the gage to the sump water level, "k" and "a" are constants determined as follows: from the data in the tables values of "a" are determined, for different discharges, using a constant value of K for all discharges, which gives a constant value of "a" for the different speeds and heads for any discharge. In order to get at the "machine loss" in the pump, calculations are made of the other losses, and their sum subtracted from "a" in the above formula, this difference being the machine loss, for the discharge considered.

The machine loss is then:

$$a - \left(\frac{v_1^2}{2g} + f_1 \frac{l_1}{d_1} \cdot \frac{v_1^2}{2g} + \frac{v_1^2}{2g} \right)$$

Taking account of all these losses, that is, taking the head pumped against in the formula for efficiency, as the sum of and a , the efficiency is seen to be approximately 81%

The 19% not yet accounted for is journal and belt friction, charged to the pump.

BIBLIOGRAPHY.

Reference has been made to the following:

Centrifugal pumps, Turbines and Water Motors, by Charles H. Innes, M. A.

Proceedings of the Institution of Civil Engineers, of London, 1878, col. 53, p. 249.

Treatise on Hydraulics. By Wm. C. Unwin.

TABLE I. TEST DATA. 800 R.P.M.

No.	Lift-Feet. Disch. Gage.	Dis. Gage to Sump level	Total	Discharge Cu. ft. per sec	Galls. per min.	R.P.M. Pump	Horsepower Delv. to Pump	Aver.	Hydraulic	Efficiency %	Aver.
1	30	10	40	1.11	500	800	14.76		5.02	34.2	
2	30	10	40	1.11	500	800	15.48	16.3	5.02	32.6	30.8
3	30	10	40	1.11	500	800	18.66		5.02	27.6	
4	50	10	60	1.96	432	800	16.57		6.56	39.7	
5	50	10	60	.96	432	800	15.26		6.56	43.0	
6	50	10	60	.96	432	800	17.88	16.6	6.56	36.7	39.5
7	50	10	60	.96	432	800	16.67		6.56	39.3	
8	70	10	80	.75	338	800	12.20		6.82	56.0	
9	70	10	80	.75	338	800	14.37		6.82	47.6	
10	70	10	80	.75	338	800	14.84	14.33	6.82	46.0	47.6
11	70	10	80	.75	338	800	15.93		6.82	42.8	
12	90	10	100	.29	130	800	11.08		3.30	29.8	
13	90	10	100	.29	130	800	9.45		3.30	35.0	
14	90	10	100	.29	130	800	9.89	9.36	3.30	33.4	35.3
15	90	10	100	.29	130	800	8.02		3.30	41.1	

TABLE II. TEST DATA. 900 R.P.M.

No.	Lift-Feet. Disch. Gage	Dis. Gage to Sump level	Total	Discharge Cu. ft. per Sec	Galls. per min	R.P.M. Pump	Horsepower Delv. to Pump	Aver.	Hydraulic	Efficiency %	Aver.
1	50	10	60	1.11	500	900	18.30		7.55	41.3	
2	50	10	60	1.11	500	900	17.73		7.55	42.6	
3	50	10	60	1.11	500	900	20.71	18.6	7.55	36.5	40.5
4	50	10	60	1.11	500	900	17.85		7.55	42.3	
5	70	10	80	.98	441	900	17.67		8.91	50.4	
6	70	10	80	.98	441	900	19.75		8.91	45.1	
7	70	10	80	.98	441	900	19.75	19.53	8.91	42.6	45.7
8	70	10	80	.98	441	900	20.95		8.91	42.6	
9	90	10	100	.78	352	900	14.97		8.87	59.3	
10	90	10	100	.78	352	900	16.78		8.87	53.0	
11	90	10	100	.78	352	900	17.77	16.47	8.87	50.0	53.8
12	90	10	100	.78	352	900	16.35		8.87	54.2	
13	110	10	120	.40	180	900	14.07		5.42	38.5	
14	110	10	120	.38	171	900	13.12		5.18	39.4	
15	110	10	120	.38	171	900	12.57	12.57	5.18	41.2	41.3
16	110	10	120	.38	171	900	12.92		5.18	43.0	

TABLE III. TEST DATA 1000 R.P.M.

No.	Lift–Feet: Disch. Gage	Lift–Feet: Dis. Gage to Sump level	Lift–Feet: Total	Discharge: Cu. ft. per Sec.	Discharge: Galls. per min	R.P.M. Pump	Horsepower: Delv. to Pump	Horsepower: Delv. to Pump Aver.	Horsepower: Hydraulic	Efficiency %:	Efficiency %: Aver.
1	20	11.9	31.9	1.61	725	1020	33.45		5.85	17.5	
2	20	11.9	31.9	1.61	725	1020	—		5.85	—	
3	20	11.9	31.9	1.61	725	1010	30.37	32.13	5.85	19.2	18.2
4	20	11.9	31.9	1.61	724	1020	32.56		5.85	17.9	
5	40	11.9	51.9	1.60	722	1020	33.16		9.46	28.5	
6	40	11.9	51.9	1.60	723	1022	31.22		9.48	30.4	
7	40	11.9	51.9	1.60	723	1023	30.68	31.95	9.48	30.8	29.7
8	40	11.9	51.9	1.60	723	1020	32.60		9.48	29.0	
9	40	11.9	51.9	1.61	724	1020	32.10		9.50	29.6	
10	60	11.9	71.9	1.53	688	1020	32.85		12.58	38.3	
11	60	11.9	71.9	1.53	687	1020	33.93		12.56	37.0	
12	60	11.9	71.9	1.53	687	1026	34.58	33.63	12.56	36.3	37.3
13	60	11.9	71.9	1.53	687	1026	33.15		12.56	37.8	
14	80	11.9	91.9	1.41	636	1024	32.00		14.78	46.1	
15	80	11.9	91.9	1.41	635	1026	32.93		14.75	44.8	
16	80	11.9	91.9	1.41	635	1026	31.48	31.42	14.75	47.0	47.1

TABLE III. cont. TEST DATA 1000 R.P.M.

No.	Lift-Feet: Disch. Gage	Lift-Feet: Dis. Gage to Sump level	Lift-Feet: Total	Discharge: Cu. ft. per Sec	Discharge: Galls. per min	R.P.M. Pump	Horsepower: Delv. to Pump	Horsepower: Delv. to Pump Aver.	Horsepower: Hydraulic	Efficiency %	Efficiency % Aver.
17	80	11.9	91.9	1.41	635	1020	29.27		14.75	50.3	
18	80	11.25	91.25	1.35	608	1026	30.57		14.05	45.8	
19	80	11.25	91.25	1.35	608	1022	29.52	29.16	14.05	47.5	48.2
20	80	11.25	91.25	1.35	605	1028	27.40		14.00	51.0	
21	100	11.25	111.25	1.13	507	1024	26.83		14.25	53.1	
22	100	11.25	111.25	1.13	507	1020	25.68		14.25	55.7	
23	100	11.25	111.25	1.13	507	1022	25.73	26.0	14.25	55.3	54.8
24	100	11.25	111.25	1.13	507	1022	25.76		14.25	55.2	
25	120	11.25	131.25	.91	412	1028	25.68		13.70	53.4	
26	120	11.25	131.25	.91	412	1030	23.33		13.70	58.5	
27	120	11.25	131.25	.92	414	1028	23.03	23.65	13.80	59.6	58.0
28	120	11.25	131.25	.91	412	1028	22.55		13.70	60.4	
29	140	11.25	151.25	.48	217	1028	17.60		8.35	47.5	
30	140	11.25	151.25	.48	214	1030	17.40	17.5	8.30	47.6	47.4
31	140	11.25	151.25	.21	94	1026	13.12		3.54	26.4	
32	140	11.25	151.25	.21	94	1026	11.84	12.5	3.54	29.8	28.3

TABLE IV. TEST DATA. 1100 R.P.M.

No.	Lift-Feet. Disch. Gage.	Dis. Gage to Sump level	Total	Discharge. Cu. ft. per Sec	Galls. per min	R.P.M. Pump	Horsepower. Delv. to Pump	Aver.	Hydraulic	Efficiency %	Aver.
1	80	10	90	1.52	683	1120	44.91		15.52	34.6	
2	80	10	90	1.53	687	1125	40.00		15.60	39.0	
3	80	10	90	1.53	687	1120	43.70	43.0	15.60	35.7	36.2
4	80	10	90	1.52	683	1120	44.50		15.53	34.8	
5	80	10	90	1.52	683	1120	41.80		15.53	37.2	
6	90	12	102	1.43	642	1120	41.10		16.60	40.4	
7	90	12	102	1.43	642	1110	37.10		16.60	44.8	
8	90	12	102	1.43	642	1110	41.40	39.1	16.60	40.3	42.3
9	90	12	102	1.43	642	1110	36.90		16.60	45.0	
10	100	12	112	1.37	617	1110	36.30		17.50	48.1	
11	100	12	112	1.37	617	1110	36.20		17.50	48.3	
12	100	12	112	1.37	617	1110	41.40	38.7	17.50	42.2	45.2
13	100	12	112	1.37	617	1110	40.80		17.50	42.8	
14	110	12	122	1.28	575	1120	36.3		17.75	48.8	
15	110	12	122	1.28	575	1120	33.6		17.75	52.8	
16	110	12	122	1.28	575	1120	35.1	35.7	17.75	50.4	49.7

TABLE IV cont. TEST DATA. 1100 R.P.M.

No.	LIFT-FEET			DISCHARGE		R.P.M.	HORSEPOWER			EFFICIENCY %	
	Disch. Gage	Dis. Gage to Sump level	Total	Cu. ft. per Sec.	Galls. per min.	Pump	Delv. to Pump	Aver.	Hydraulic		Aver.
17	110	12	122	1.28	575	1120	37.90		17.75	46.7	
18	120	10	130	1.20	540	1120	37.60		17.70	47.1	
19	120	10	130	1.21	544	1120	35.30		17.80	50.3	
20	120	10	130	1.21	544	1120	38.00	37.2	17.80	46.8	47.8
21	120	10	130	1.21	544	1120	37.95		17.80	46.9	
22	130	10	140	1.13	508	1120	35.40		18.00	51.0	
23	130	10	140	1.13	508	1120	35.40		18.00	51.0	
24	130	10	140	1.13	508	1120	35.30	35.0	18.00	51.1	51.3
25	130	10	140	1.13	508	1120	34.00		18.00	53.0	
26	140	10	150	1.08	485	1120	36.60		18.50	50.6	
27	140	10	150	1.08	485	1120	32.45		18.50	57.0	
28	140	10	150	1.08	485	1120	32.40	34.2	18.50	57.1	54.1
29	140	10	150	1.08	485	1120	35.30		18.50	52.2	
30	150	10	160	.98	440	1120	30.80		17.85	57.8	
31	150	10	160	.98	440	1120	35.30		17.85	50.7	
32	150	10	160	.98	440	1120	32.90	33.4	17.85	54.2	53.6

TABLE IV, cont. TEST DATA. 1100 R.P.M.

No.	Lift-Feet. Disch. Gage	Dis. Gage to Sump level	Total	Discharge Cu. ft. per Sec.	Galls. per min.	R.P.M. Pump	Horsepower Delv. to Pump	Aver.	Hydraulic	Efficiency %	Aver.
33	150	10	160	.98	440	1120	34.70		17.85	51.4	
34	160	10	170	.85	383	1120	31.50		16.45	52.2	
35	160	10	170	.85	383	1120	31.20		16.45	52.8	
36	160	10	170	.85	383	1120	32.55	31.8	16.45	50.4	51.8
37	160	10	170	.85	383	1120	32.10		16.45	51.3	
38	170	10	180	.68	305	1120	27.25		13.95	51.2	
39	170	10	180	.68	305	1120	27.80		13.95	50.1	
40	170	10	180	.68	305	1120	30.70	28.5	13.95	45.5	49.0
41	170	10	180	.68	305	1120	28.20		13.95	49.5	
42	180	10	190	.43	194	1120	22.20		9.27	41.8	
43	180	10	190	.43	194	1120	21.20		9.27	43.8	
44	180	10	190	.43	194	1120	21.30	22.1	9.27	43.6	42.0
45	180	10	190	.43	194	1120	23.80		9.27	39.0	

Table V. Test Data. 1200 R.P.M.

No.	Lift-Feet: Disch. Gage.	Lift-Feet: Dis. Gage to Sump level	Lift-Feet: Total	Discharge: Cu. ft. per Sec.	Discharge: Galls. per min.	R.P.M. Pump	Horsepower: Delv. to Pump	Horsepower: Delv. to Pump Aver.	Horsepower: Hydraulic	Efficiency %	Efficiency % Aver.
1	90	10	100	1.61	725	1190	45.8		18.3	39.8	
2	90	10	100	1.61	725	1190	48.9		18.3	37.3	
3	90	10	100	1.61	725	1190	47.2	47.6	18.3	38.6	38.4
4	90	10	100	1.61	725	1190	48.6		18.3	37.5	
5	110	10	120	1.51	675	1190	46.3		20.5	44.3	
6	110	10	120	1.51	675	1190	44.9		20.5	45.8	
7	110	10	120	1.51	675	1190	44.1	45.2	20.5	46.5	45.4
8	110	10	120	1.51	675	1190	45.4		20.5	45.2	
9	130	10	140	1.39	621	1190	44.0		22.0	50.0	
10	130	10	140	1.39	621	1190	45.4		22.0	48.5	
11	130	10	140	1.39	621	1190	43.2	43.6	22.0	51.0	50.3
12	130	10	140	1.39	621	1190	41.8		22.0	52.6	
13	150	10	160	1.24	558	1190	42.1		22.6	53.5	
14	150	10	160	1.24	558	1190	43.0		22.6	52.5	
15	150	10	160	1.24	558	1190	40.9	41.8	22.6	55.2	54.0
16	150	10	160	1.24	558	1190	41.2		22.6	54.8	

TABLE V. cont. TEST DATA. 1200 R.P.M.

No.	LIFT-FEET.			DISCHARGE		R.P.M.	HORSEPOWER			EFFICIENCY %	
	Disch. Gage.	Dis. Gage to Sump level	Total	Cu. ft. per Sec.	Galls. per min.	Pump.	Delv. to Pump	Aver.	Hydraulic		Aver.
17	170	10	180	1.10	495	1190	40.5		22.5	55.5	
18	170	10	180	1.10	495	1190	38.2		22.5	59.0	
19	170	10	180	1.10	495	1190	37.4	38.8	22.5	60.2	58.1
20	170	10	180	1.10	495	1190	39.2		22.5	57.3	
21	190	10	200	.91	410	1190	37.6		21.6	57.2	
22	190	10	200	.91	410	1190	37.6		21.6	57.2	
23	190	10	200	.91	410	1190	35.2	36.0	21.6	61.2	60.1
24	190	10	200	.91	410	1190	33.6		21.6	64.0	

TABLE VI TEST DATA HEAD - 59 Ft.

No.	Lift-Feet: Disch. Gage	Lift-Feet: Dis. Gage to Sump level	Lift-Feet: Total	Discharge: Cu.ft. per Sec.	Discharge: Galls. per min	R.P.M. Pump	Horsepower: Delv. to Pump	Horsepower: Delv. to Pump Aver.	Horsepower: Hydraulic	Efficiency %	Efficiency % Aver.
1	50	9.25	59.25	1.35	608	878	—		9.05	—	
2	50	9.25	59.25	1.30	584	876	38.9		8.80	22.7	
3	50	9.25	59.25	1.32	594	880	26.6		8.95	33.5	
4	50	9.25	59.25	1.34	596	908	25.8	28.2	8.96	34.5	31.7
5	50	9.25	59.25	1.34	596	894	24.6		8.96	36.2	
6	50	9.25	59.25	1.33	595	892	25.1		8.98	35.5	
7	50	9.25	59.25	1.52	675	960	29.5		10.10	34.2	
8	50	9.25	59.25	1.69	758	1032	38.1		11.40	29.8	
9	50	9.25	59.25	1.67	756	1042	36.0		11.40	31.6	
10	50	9.25	59.25	1.71	765	1032	35.9		11.55	32.2	
11	50	9.25	59.25	1.70	763	1038	34.8	35.8	11.54	33.2	32.1
12	50	9.25	59.25	1.71	765	1034	36.8		11.55	31.4	
13	50	9.25	59.25	1.69	761	1030	33.5		11.52	34.3	

TABLE VII.

LOST HEAD.

Values for Constants.

DISCHARGE Galls. per min	k	a	$n = k\sqrt{h+a}$
200	77	10	$n = 77\sqrt{h+10}$
250	77	15	$n = 77\sqrt{h+15}$
300	77	25	$n = 77\sqrt{h+25}$
350	77	35	$n = 77\sqrt{h+35}$
400	77	45	$n = 77\sqrt{h+45}$
440	77	55	$n = 77\sqrt{h+55}$
500	77	75	$n = 77\sqrt{h+75}$
600	77	85	$n = 77\sqrt{h+85}$
680	77	105	$n = 77\sqrt{h+105}$

Revolution Table.

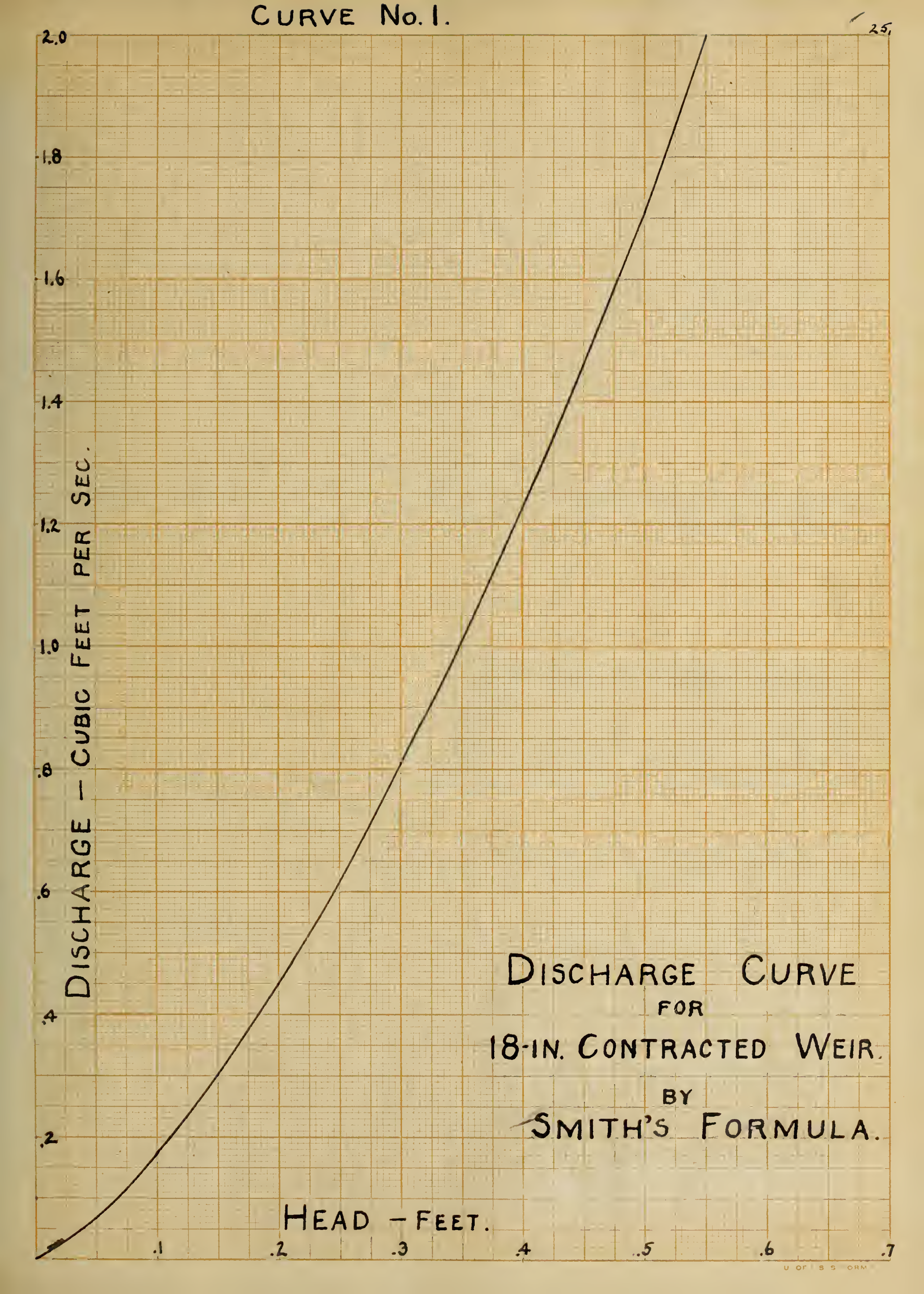
Table VIII.

Head	Discharge – Galls. per min.								
Feet	200	250	300	350	400	450	500	600	700
50	570	620	670	710	750	790	860	895	975
75	685	730	770	808	845	880	945	975	1050
100	790	825	860	900	930	965	1020	1050	1115
125	875	915	940	970	1005	1040	1090	1115	1185
150	965	990	1020	1050	1080	1110	1160	1185	1245
175	1035	1065	1090	1120	1145	1170	1220	1245	1310
200	1100	1130	1160	1185	1205	1230	1280	1310	1360

TABLE IX.

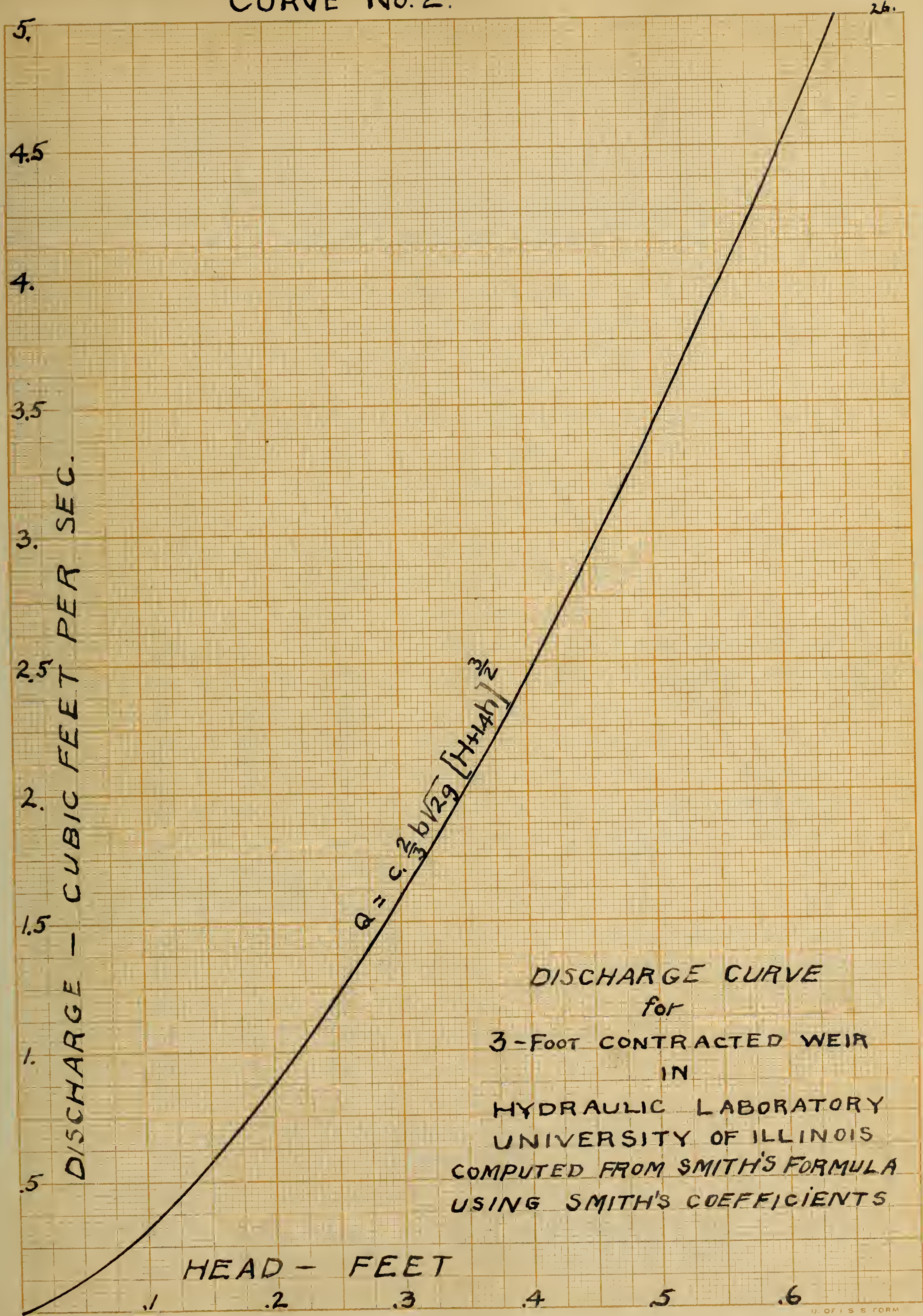
Losses.

Discharge		Entrance Head	Entrance To Suction	Friction in Suction	Discharge Head	Machine Loss.
Cu.ft. per Sec	Galls. per min.	Feet	Feet	Feet	Feet	Feet
.78	350	.502	.502	.276	1.270	32.45
.98	440	.786	.786	.415	2.010	51.003
1.11	500	1.010	1.010	.530	2.580	69.870
1.33	600	1.470	1.470	.740	3.720	77.600
1.51	680	1.890	1.890	.910	4.810	95.500

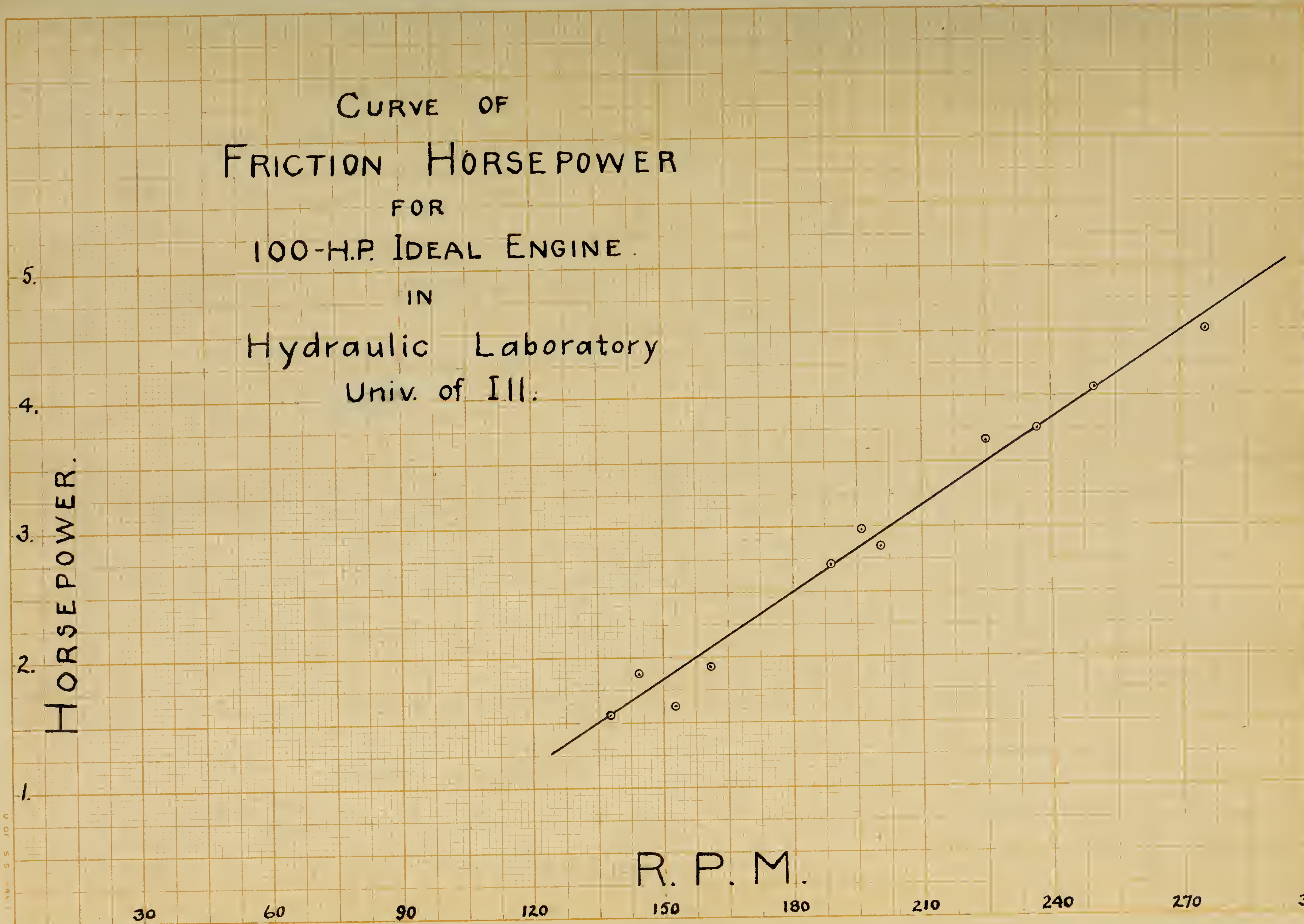


CURVE No. 1.
25
2.0
1.8
1.6
1.4
1.2
1.0
.8
.6
.4
.2
DISCHARGE - CUBIC FEET PER SEC.
DISCHARGE CURVE
FOR
18-IN. CONTRACTED WEIR
BY
SMITH'S FORMULA.
HEAD - FEET.
.1
.2
.3
.4
.5
.6
.7

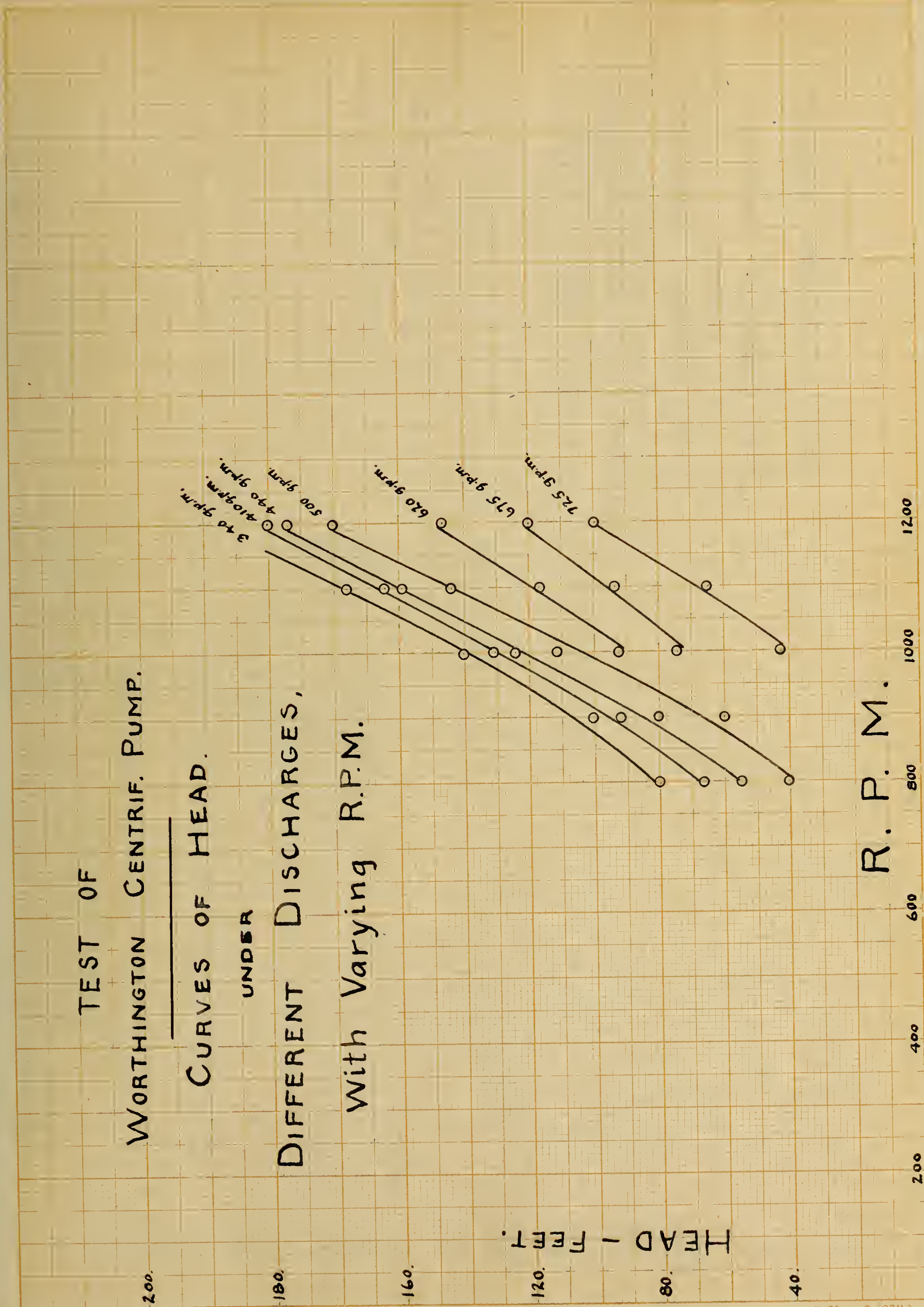
CURVE No. 2.

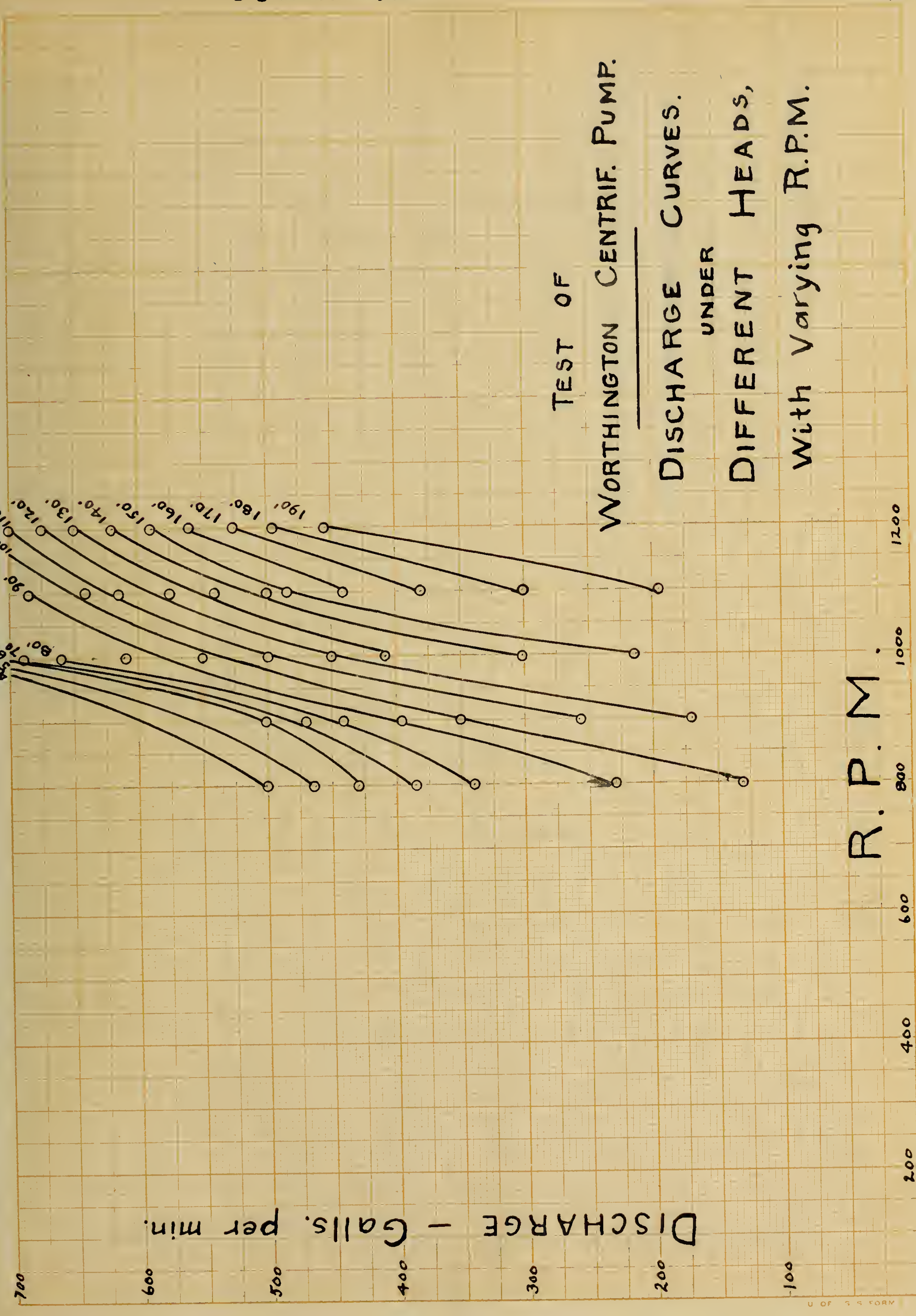

26.
5.
4.5
4.
3.5
3.
2.5
2.
1.5
1.
.5
DISCHARGE - CUBIC FEET PER SEC.
$Q = c\,\frac{2}{3}\,b\sqrt{2g}\,[H+1.4h]^{\frac{3}{2}}$
DISCHARGE CURVE
for
3-FOOT CONTRACTED WEIR
IN
HYDRAULIC LABORATORY
UNIVERSITY OF ILLINOIS
COMPUTED FROM SMITH'S FORMULA
USING SMITH'S COEFFICIENTS
HEAD - FEET
.1
.2
.3
.4
.5
.6
U. OF I. S. S. FORM

CURVE No. 3.

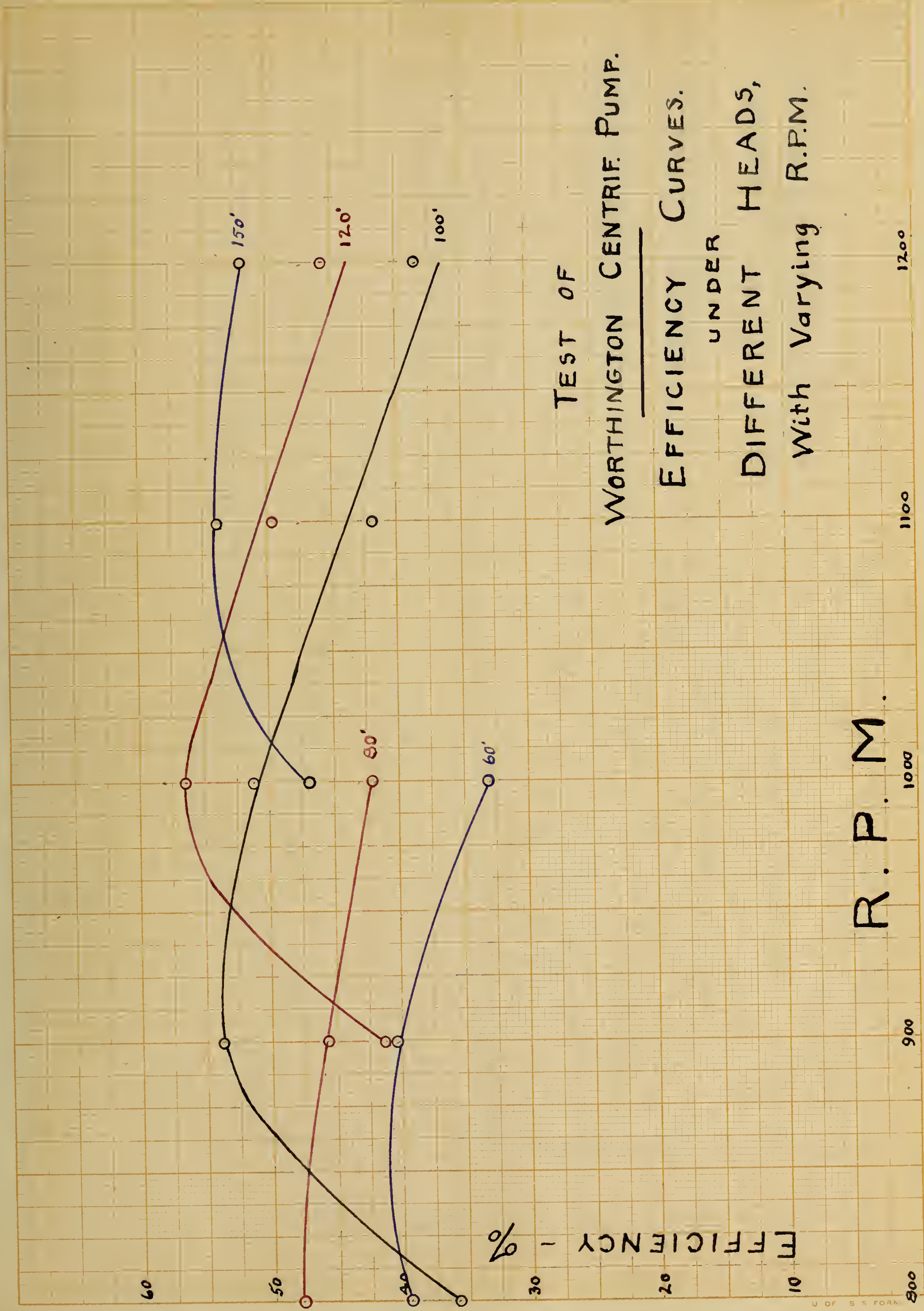

CURVE OF
FRICTION HORSEPOWER
FOR
100-H.P. IDEAL ENGINE
IN
Hydraulic Laboratory
Univ. of Ill.
HORSEPOWER.
R. P. M.
1.
2.
3.
4.
5.
30
60
90
120
150
180
210
240
270
300

CURVE No. 4.


TEST OF
WORTHINGTON CENTRIF. PUMP.
CURVES OF HEAD.
UNDER
DIFFERENT DISCHARGES,
With Varying R.P.M.
340 g.p.m.
410 g.p.m.
490 g.p.m.
500 g.p.m.
620 g.p.m.
675 g.p.m.
725 g.p.m.
R. P. M.
200
400
600
800
1000
1200
HEAD - FEET.
200.
180.
160.
120.
80.
40.

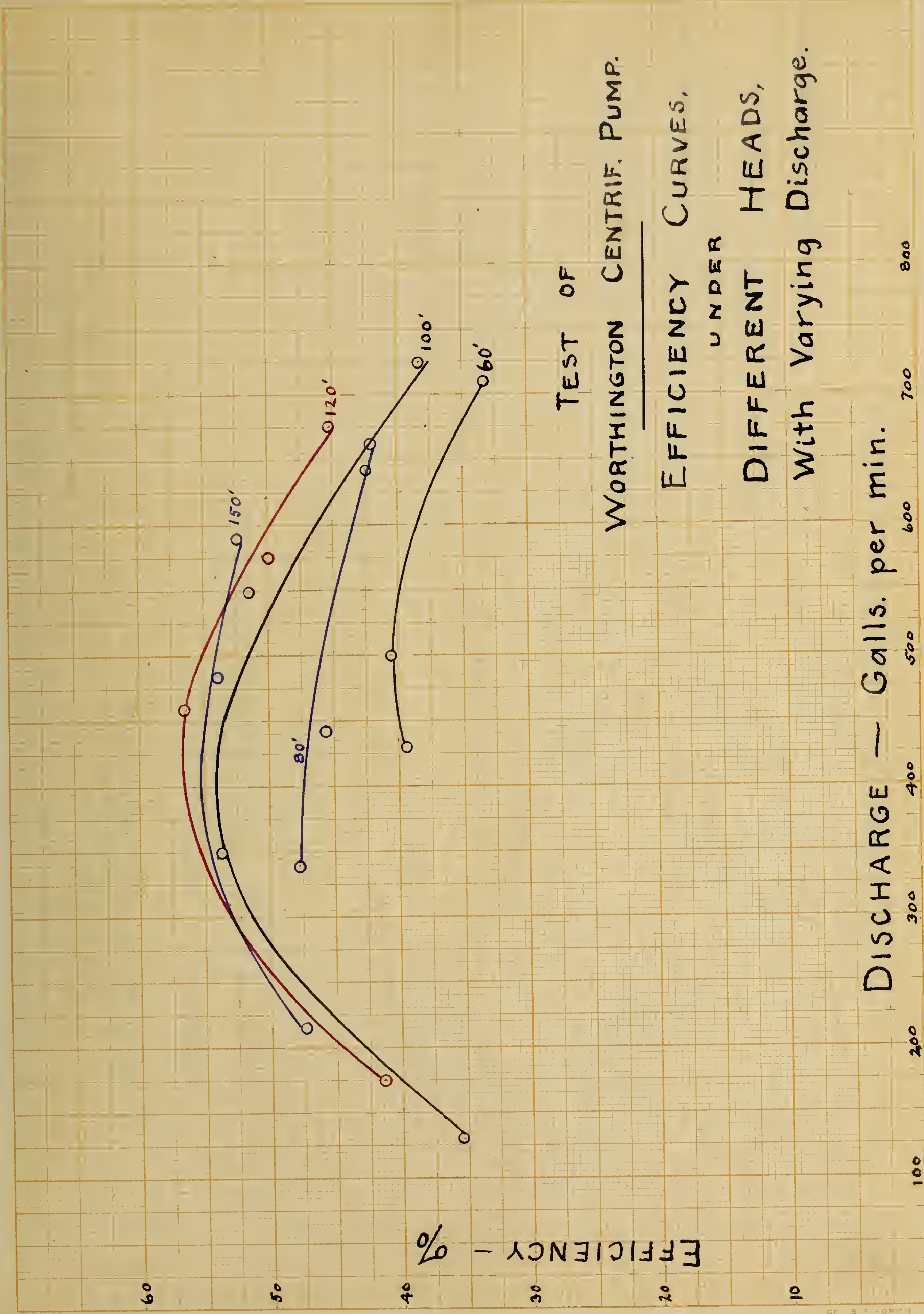

TEST OF
WORTHINGTON CENTRIF. PUMP.
DISCHARGE CURVES.
UNDER
DIFFERENT HEADS,
With Varying R.P.M.
R. P. M.
DISCHARGE — Galls. per min.
200
400
600
800
1000
1200
100
200
300
400
500
600
700
40'
50'
60'
70'
80'
90'
100'
110'
120'
130'
140'
150'
160'
170'
180'
190'

CURVE No. 6.

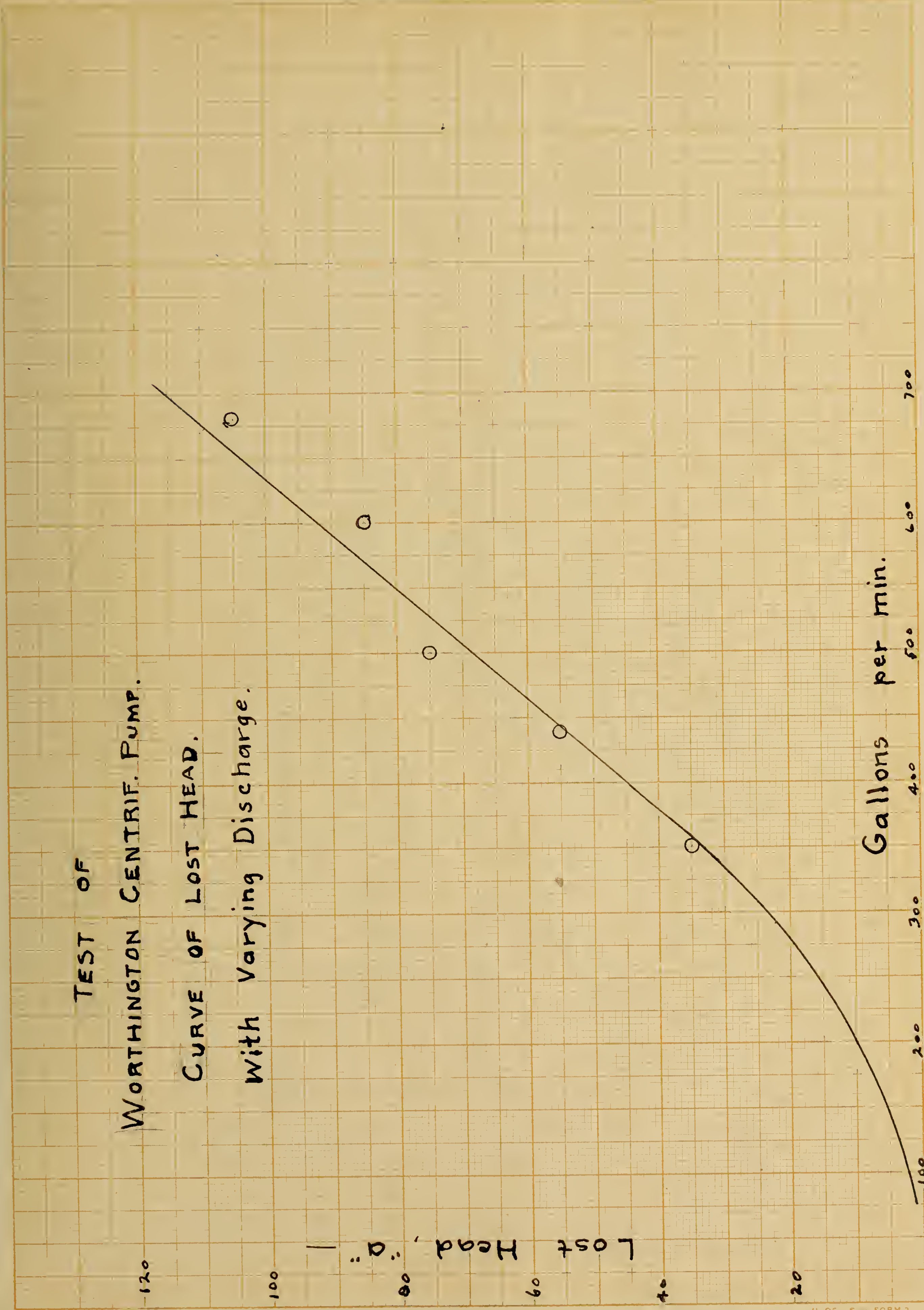


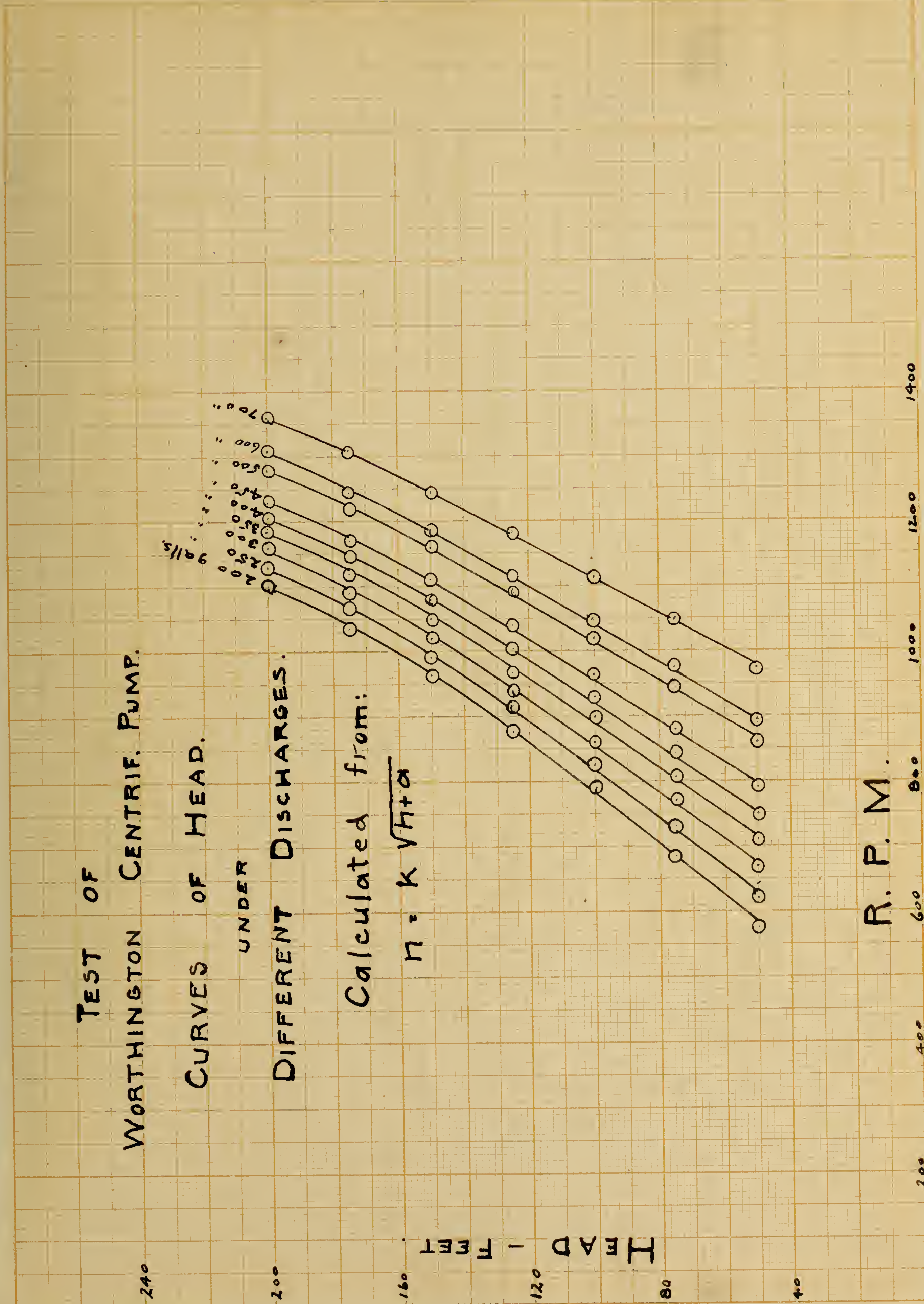
TEST OF
WORTHINGTON CENTRIF. PUMP.
EFFICIENCY CURVES.
UNDER
DIFFERENT HEADS,
With Varying R.P.M.
150'
120'
100'
80'
60'
R. P. M.
EFFICIENCY - %
60
50
40
30
20
10
800
900
1000
1100
1200

CURVE No. 7.


TEST OF
WORTHINGTON CENTRIF. PUMP.
EFFICIENCY CURVES,
UNDER
DIFFERENT HEADS,
With Varying Discharge.
DISCHARGE — Galls. per min.
EFFICIENCY – %
150'
120'
100'
80'
60'
100
200
300
400
500
600
700
800
10
20
30
40
50
60

CURVE No. 9.


TEST OF
WORTHINGTON CENTRIF. PUMP.
CURVE OF LOST HEAD.
With Varying Discharge.
Gallons per min.
100
200
300
400
500
600
700
Lost Head, "a".
20
40
60
80
100
120
U OF S. G. FORM 7


TEST OF
WORTHINGTON CENTRIF. PUMP.
CURVES OF HEAD.
UNDER
DIFFERENT DISCHARGES.
Calculated from:
$n = K\sqrt{H+a}$
R. P. M.
HEAD - FEET
200 galls.
250
300
350
400
450
500
600
700
240
200
160
120
80
40
200
400
600
800
1000
1200
1400